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A Heat Receiver Design for Solar Dynamic Space Power Systems

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ABSTRACT

This paper describes an advanced heat pipe heat receiver designed for a solar dynamic space power system. The power system consists of a solar concentrator, solar heat receiver, Stirling heat engine, linear alternator and waste heat radiator. The solar concentrator focuses the sun's energy into a heat receiver. The engine and alternator convert a portion of this energy to electric power and the remaining heat is rejected by a waste heat radiator. Primary liquid metal heat pipes transport heat energy to the Stirling engine. Thermal energy storage allows this power system to operate during the shade portion of an orbit. Lithium fluoride/calcium fluoride eutectic is the thermal energy storage material. Thermal energy storage canisters are attached to the midsection of each heat pipe. The primary heat pipes pass through a secondary vapor cavity heat pipe near the engine and receiver interface. The secondary vapor cavity heat pipe serves three important functions. First, it smooths out hot spots in the solar cavity and provides even distribution of heat to the engine. Second, in the event of a heat pipe failure, the secondary heat pipe cavity can efficiently transfer heat from other operating primary heat pipes to the engine heat exchanger of the defunct heat pipe. Third, the secondary heat pipe vapor cavity reduces temperature drops caused by heat flow into the engine. This unique design provides a high level of reliability and performance.

INTRODUCTION

The NASA Lewis Research Center has produced a conceptual design of a 1.5 kWe (2.0 hp) solar dynamic space power system. The primary goal for this project was to demonstrate the feasibility of solar dynamic power systems in space. To date, solar dynamic space power systems have not been used in space. Studies have shown that for some applications solar dynamic power systems offer distinct advantages over other space power systems. For example, solar dynamic power is being considered for use on later phases of Space Station Freedom. If solar dynamic power systems can be proven in space, future space missions can benefit from their use.

DESIGN REQUIREMENTS

The design requirements for the heat receiver were:

(1) The heat receiver must have high reliability.

(2) Single point failures must be minimized.

(3) The heat receiver must provide uniform heat to the Stirling engine heater head.

(4) The heat receiver must provide heat to the Stirling engine in both sun and shade portions of an orbit.

(5) The heat receiver must tolerate start-up and shut-down transients.

GENERAL DESCRIPTION

A heat receiver design developed by Allied Signal Aerospace Company under a NASA contract (Strumpf and Coombs, 1987) for the Advanced Solar Dynamic Program at the NASA Lewis, served as a basis for the design described in this paper. Two features of the Allied Signal design are used in this particular heat receiver design. These features are:

(1) The Thermal Energy Storage (TES) material is placed in small canisters to tolerate void formation and improve heat conduction. Voids are created by the large volume change when solidification occurs. Voids are distributed throughout the volume of TES material by placing material in small canisters. The walls of the TES canisters improve heat conduction into and out of the TES material by acting as fins.

(2) The primary heat pipes transport thermal energy between the solar cavity, the TES material, and the Stirling engine.

However; the method of absorbing solar energy by the primary heat pipes and attaching the receiver to the engine are entirely different from the Allied Signal design.

A schematic of the solar dynamic heat receiver coupled to the Stirling engine is shown in Fig. 1. The three sections of the receiver are: a solar cavity, a thermal energy storage section and a heat pipe cavity and engine interface section. There are two different types of heat pipes in the receiver, primary and secondary. Primary heat pipes extend the length of the receiver. Canisters of TES material are slipped over

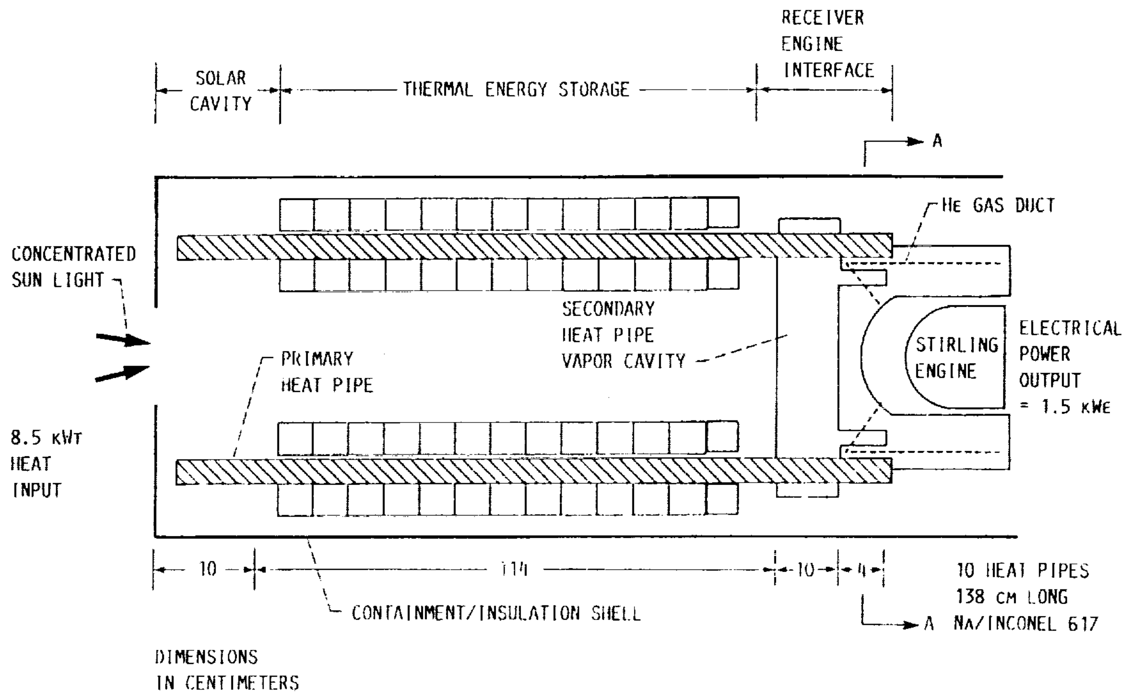

FIGURE 1. - SCHEMATIC OF A SOLAR DYNAMIC HEAT RECEIVER.

and brazed to them in the TES section. The primary heat pipes pass through the secondary heat pipe vapor cavity to the engine. The secondary heat pipe vapor cavity thermally interconnects all of the primary heat pipes near the engine, as shown in Figs. 2 and 3. The secondary heat pipe cavity ensures uniform heat input into the engine as well as improving the reliability of the receiver.

The majority of the receiver and engine is made of Inconel 617. The heat pipe working fluid is sodium. Heat pipe wicks are made from several layers of Inconel screen. High conductivity nickel fin sections are inserted and brazed into the engine heater head. These fins reduce the temperature drop from the heat pipes to the engine working gas (Fig. 3).

RECEIVER OPERATION

Solar energy reflected by the concentrator enters through the aperture of the receiver during the sun portion of a planetary orbit. The evaporator section of the primary heat pipes and the TES canisters absorb most of the energy that comes through the aperture. Part of the energy that enters the receiver is stored in the TES material and the remaining portion is transported to the Stirling engine, with minor heat losses to space.

The secondary heat pipe vapor cavity performs three functions:

(1) It provides the engine with a uniform heat flux. If the solar energy entering the receiver is not uniform, some of the primary heat pipes will operate at higher temperatures and transport more heat than other primary heat pipes. The secondary heat pipe vapor cavity will transport heat from primary heat pipes at higher temperatures to primary heat pipes at lower temperatures. This will provide the engine with a uniform input heat distribution.

(2) It allows the engine to efficiently operate with several primary heat pipe failures. As shown in Figs. 2 and 3, the secondary heat pipe vapor cavity extends into the heater head of the Stirling engine.

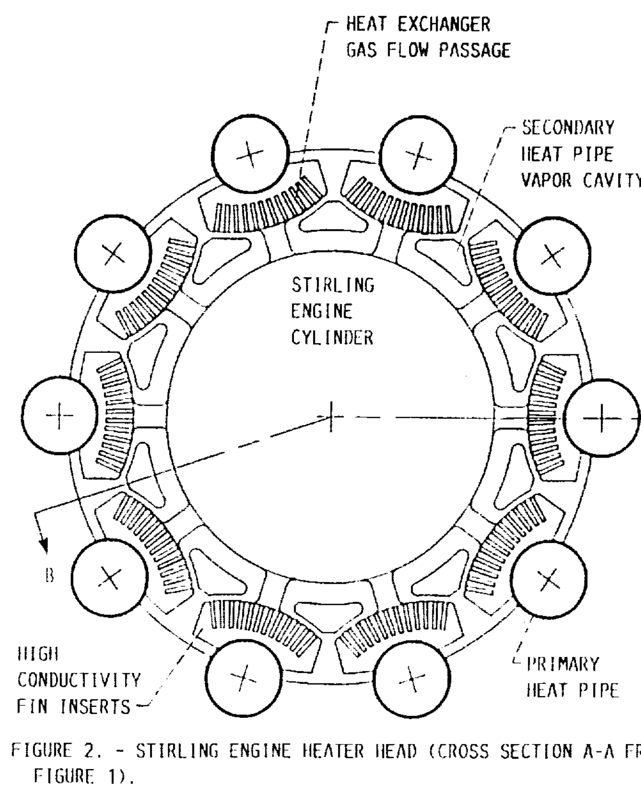

FIGURE 2. - STIRLING ENGINE HEATER HEAD (CROSS SECTION A-A FROM FIGURE 1).

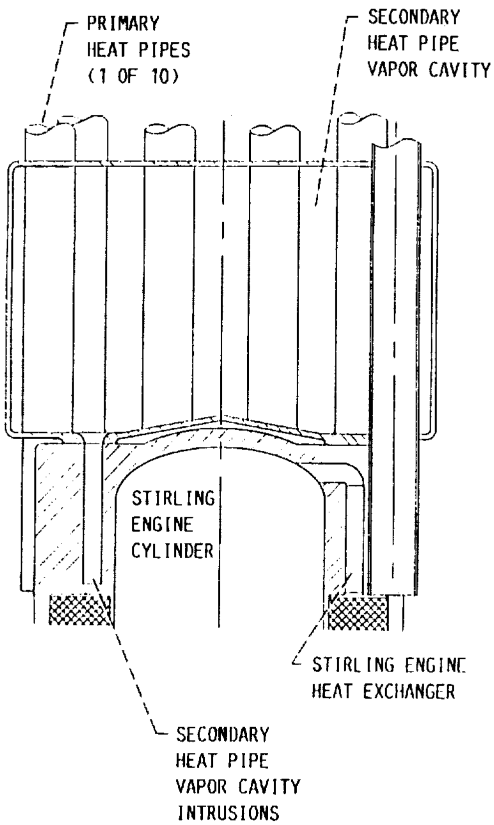

FIGURE 3. - HEAT RECEIVER STIRLING ENGINE INTERFACE (SECTION B-B FROM FIGURE 2).

In the event of a primary heat pipe failure, the secondary heat pipe vapor cavity will transfer heat from operating primary heat pipes to the engine heat exchanger of the defunct primary heat pipe. This will provide a uniform input heat distribution to the engine with a primary heat pipe failure.

(3) It reduces the temperature drop associated with heat transfer into the engine by providing a second parallel path. A finite element analysis has shown that the heat pipe vapor cavity reduces the temperature drop by about half.

At start-up, solar energy will directly heat both the primary heat pipe evaporators in the solar cavity section and the outside of the TES canisters in the thermal energy storage section, see Fig. 1. Calculations indicate that operating conditions will be achieved after three orbits. Care has been exercised to ensure that the TES operates in a partially melted condition at all times. This is needed to prevent excessive temperature fluctuations.

RECEIVER DESIGN

Since solar receiver and Stirling engine development are generally carried out by distinct organizations, there has been a tendency to optimize each component individually and to disregard certain interface problems. The effort undertaken at NASA Lewis has permitted the two design groups to operate within a single team so that the appropriate trade-offs were made. It became apparent in the early stages of this effort that the receiver design tended to optimize at larger diameters relative to the engine. Temperature drops at the heat exchanger of the engine were reduced because of the larger heat transfer area associated with the large receiver diameter. Also a large receiver diameter can accommodate the TES material with a short receiver length. From a packaging standpoint, it is desired to minimize the receiver length. The engine, on the other hand, tended to optimize at small diameters relative to the receiver. Small diameter engines operate at high pressure with low gas volume which typically yields high efficiencies. A trade-off study was performed to determine the engine diameter that provided reasonable receiver packaging without compromising engine efficiency. The study showed that at the 1.5 kWe (2.0 hp) power level the performance of the engine and the receiver are not overly sensitive to these dimensional changes.

A number of receiver design variations of this concept were considered. The parameters varied between designs were: the number of primary heat pipes and primary heat pipe diameter. The optimized Stirling engine heater head diameter was found to be 13.3 cm (5.2 in.). After several iterations it was decided to use straight primary heat pipes instead of bent. The thermal energy storage canister o.d. was sized so that each canister touched adjacent canisters. The heat receiver design charactistics are shown in Table I. Some of the results of these calculations are shown in Table II. Note that as the number of primary heat pipes increase, the o.d. of the TES canisters decrease. This occurs because the spacing between primary heat pipes decreases with more heat pipes around the same perimeter. Since the o.d. of the TES canisters decreases each canister contains less material and since the amount of TES material needed is constant, the number of TES canisters increases. This results in an increased receiver length.

The use of smaller diameter primary heat pipes increases the spacing between primary heat pipes. Thus each TES canister will hold more material which tends to reduce the length of the heat receiver. On the negative side, temperature variation during an orbit is increased because the radial thickness of the TES is increased and the heat transfer area at the inside of the TES canister is reduced. While this is undesirable, the temperature variation for this design is small and some variation is tolerable. A more important reason to avoid thermal variations is the associated thermal stresses which may cause failure. The smaller diameter heat pipes, 1.3 cm o.d. (0.5 in.), produced another negative effect. The heat pipes complicate the design of the gas passages in the engine heater head and decrease the azimuthal uniformity of heat input to the engine.

The final design specifications for the heat receiver design are shown in Table III.

ANALYSIS

The analysis which follows was used to establish the performance of the final design. As shown in Table III, the final configuration used 10, 1.9 cm o.d. (0.75 in.) heat pipes with 4.1 cm o.d. (1.6 in.) TES canisters.

The analysis was conducted with a variety of codes. Heat pipe limits were determined with the NASA Lewis steady state heat pipe code (Baker and Tower, 1988). The primary heat pipes operate well below the boiling, capillary, sonic and entrainment limits. Boiling in the wick due to uneven heat flux distribution along the primary heat pipe evaporator section in the solar cavity was investigated. The peak heat flux will occur at a point on the primary heat pipe that intercepts the

TABLE I. - HEAT RECEIVER DESIGN CHARACTERISTICS

Characteristic	Size
Primary heat pipe bolt circle, cm (in.)	13.3 (5.2)
Primary heat pipe wall thickness, cm (in.)	0.056 (0.022)
TES canister inside wall thickness, cm (in.)	0.071 (0.028)
TES canister side wall thickness, cm (in.)	0.15 (0.060)
TES canister outside wall thickness, cm (in.)	0.071 (0.028)
Engine thermal input power, kWh (Btu/hr)	4.95 (16890)
Solar cavity radiative losses, kWh (Btu/hr)	1.22 (4.16)
Receiver side wall losses, kWh (Btu/hr)	0.08 (0.273)
TES storage over-design	1.2
TES material density, kg/m^3 (lbm/ft^3)	2097 (130.9)
TES material heat of fusion, kJ/kg (Btu/lbm)	753 (324)

TABLE II - HEAT RECEIVER DESIGN TRADE-OFFS

[Heat pipe length versus the number of heat pipes.]

Number of primary heat pipes	TES canister outside diameter, cm (in.)	TES section length, cm (in.)	Number of TES canisters
8	5.08 (2.00)	78.7 (31.0)	440
9	4.55 (1.79)	94.0 (37.0)	583
10	4.09 (1.61)	111.8 (44.0)	792
11	3.73 (1.47)	134.6 (53.0)	1061
12	3.43 (1.35)	167.6 (66.0)	1274

TABLE III - HEAT RECEIVER DESIGN SPECIFICATIONS

Number of primary heat pipes	10
Length of primary heat pipes, cm (in.)	138 (54.3)
Primary heat pipe inside diameter, cm (in.)	1.9 (0.75)
Heat pipe envelope material	Inconel
Heat pipe working fluid	Sodium
TES canister outside diameter, cm (in.)	4.06 (1.60)
TES canister material	Inconel
TES material	LiF-22CaF2
TES section length	114 (44.9)
Number of TES canisters	792
Heat input to TES, day, kWth (Btu/hr)	4.71 (16 071)
Heat output of TES, night, kWth(Btu/hr)	5.06 (17 266)
Sun time per orbit, min	60
Eclipse time per orbit, min	36

solar flux reflected from the outer perimeter of the concentrator. The CAV2 code, developed by Sandia National Labs (Diver and Andraka, 1987) was used for this calculation. The code accounts for the concentrator and receiver geometric relationship. Figure 4 shows the results of this calculation. The fraction of incident energy is plotted against the axial distance from the aperture. The peak flux shown is 16 percent of the total energy entering the receiver. The maximum radial heat flux at this point was calculated to be less than 30 W/cm^2 (660 $Btu/hr/in.^2$). This heat flux would occur over only a small area of the primary heat pipe. The actual value depends largely on the effective absorbtivity and emissivity of the Inconel heat pipe evaporators.

The SOLDTESS code was developed at the University of South Florida (Crane, 1989) to determine the temperature of the primary heat pipes during an orbit. The vapor temperature of the primary heat pipes was found to be 1039±4 K (1411±7.2 °F) as shown in Fig. 5. The magnitude of this variation is low compared to previous designs. This is attributed to the relatively small outside diameter of the TES canisters.

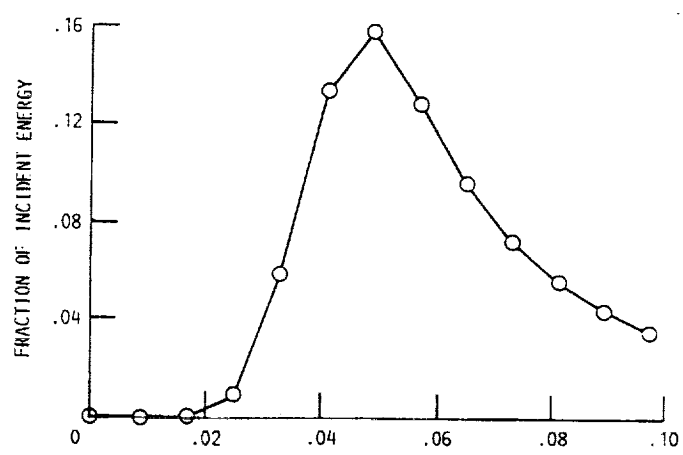

FIGURE 4. - INCIDENT SOLAR FLUX DISTRIBUTION ON PRIMARY HEAT PIPE EVAPORATORS.

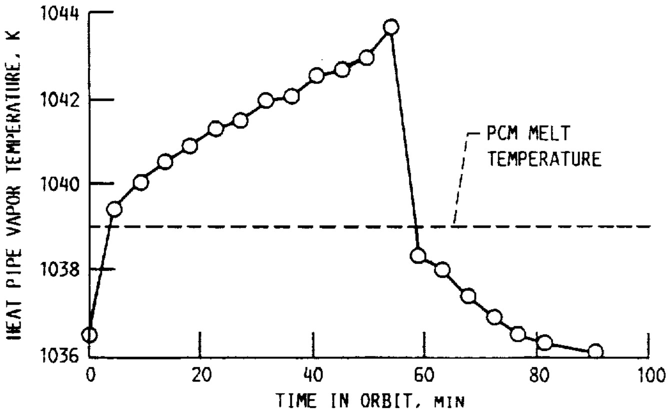

FIGURE 5. - PRIMARY HEAT PIPE VAPOR TEMPERATURE VERSUS TIME IN ORBIT.

The GLIMPS code was developed by Gedeon Associates (Gedeon, 1986) and used to perform an analysis on the engine heater head. The code calculates Stirling engine performance for specific engine designs. Calculations indicated that a average mass flow of 0.343 kg/s (0.76 lbm/s) and a average heat transfer convection coefficient of 1.227 kW/m^2 K (0.756 Btu/hr/ft 2/°F) produce a thermal condition at the engine heat exchanger equivalent to the operating engine. The heater head heat exchanger has 120 gas passages with dimensions of 1 by 6.7 by 40 mm (0.04 by 0.26 by 1.58 in.). A cross section of the engine heater head is shown in Fig. 2. The primary heat pipes are shown along with the somewhat triangular intrusions of the secondary heat pipe cavity. A finite element analysis showed that under normal operating conditions 66 percent of the heat entering the engine working fluid does so by means of the primary heat pipes. The remaining 34 percent enters by means of the intrusions of the secondary heat pipe cavity, shown in Fig. 3.

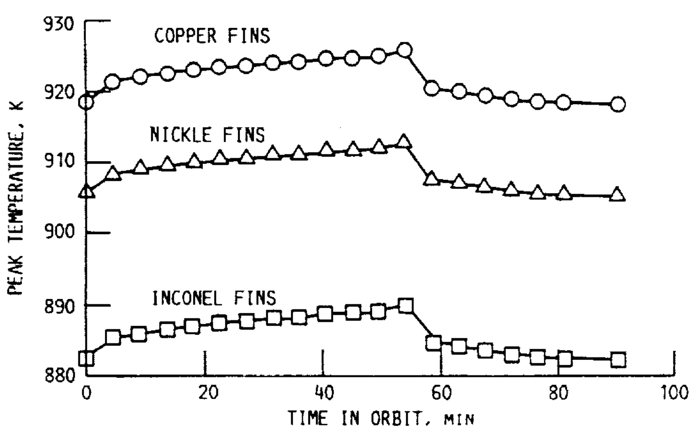

FIGURE 6. - ENGINE PEAK GAS TEMPERATURE VERSUS TIME IN ORBIT.

Several materials were evaluated for use as high conductivity fin inserts in the heat exchanger of the engine heater head shown in Fig. 2. Nickel was chosen as the best material. The engine peak temperature was 23 K (41 °F) higher with the nickel insert than it was with Inconel fins as shown in Fig. 6. Copper inserts would allow the engine to operate about 36 K (65 °F) higher than with Inconel fins as shown in Fig. 6. However, copper was not chosen because of its low strength at the operating temperatures for this design.

CONCLUDING REMARKS

NASA Lewis has designed a 1.5 kWe (2.0 hp) solar dynamic space power system. The heat receiver design for this system has been outlined in this paper. The design is different from previous designs because of constraints imposed by interfacing it with a Stirling engine and by mission requirements. The heat receiver can operate efficiently with component failures and provide the Stirling engine with uniform heat flow at the heater head at all times.

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15. Supplementary Notes

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16. Abstract

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